Notice No.1

for the
Code for Lifting Appliances in a Marine Environment
July 2016

The status of this Rule set is amended as shown and is now to be read in conjunction with this and prior Notices. Any corrigenda included in the Notice are effective immediately.

Issue date: August 2016

Amendments to	**Effective date**
Chapter 1, Section 1	Corrigenda
Chapter 1, Section 3	1 September 2016
Chapter 2, Sections 2 & 7	Corrigenda
Chapter 3, Section 1	Corrigenda
Chapter 4, Sections 2 & 3	Corrigenda
Chapter 4, Section 5	1 September 2016
Chapter 5, Section 5	Corrigenda
Chapter 6, Section 2	Corrigenda
Chapter 7, Section 2	1 September 2016
Chapter 8, Section 2	Corrigenda
Chapter 8, Sections 3 & 5	1 September 2016
Chapter 8, Sections 3 & 5	Corrigenda
Chapter 9, Sections 5 & 6	1 September 2016
Chapter 12, Sections 1 & 3	1 September 2016
Chapter 13, Section 1	Corrigenda

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Chapter 1
General

■ Section 1
Introduction

1.3 Classification

Table 1.1.1 Special features class notations associated with lifting appliances.

Lifting appliance	Special features class notation	Remarks
Cargo gear, derricks, derrick cranes or cranes on ships	**CG**	Optional notation Indicates that the ship's cargo gear is included in class
Units, offshore installations and offshore support vessels	**OC**	Optional notation Indicates that the installation's unit's or vessel's main deck cranes are included in class
Pipe laying system	**PLS**	Optional notation Indicates that the pipe laying system is included in class
~~Personal~~ Personnel transfer system (Walk-to-Work)	**W2W**	Optional notation Indicates that the ~~personal~~ personnel transfer system is included in class
Lifts and ramps on ships	**CL** **PL** **CR**	Optional notations Indicate that the ship's cargo lifts (CL), passenger lifts (PL) or cargo ramps (CR) are included in class
Lifting appliances fitted to military naval ships	**LAP**	Optional notation Indicates that lifting appliances have been designed and built in accordance with this Code (or equivalent standard), but where it is not mandated by the LA notation
Lifting appliances forming an essential feature of the vessel, e.g. cranes on crane barges or pontoons, some lifting arrangements for diving on diving support ships, etc.	**LA**	Mandatory notation Indicates that the lifting appliance is included in class

■ Section 3
Plans and information to be submitted

3.6 Hydraulic cylinders

3.6.1 Fully detailed and dimensioned engineering drawings are to be submitted for appraisal, indicating the length, thicknesses, diameters and material specifications of the main elements, including cylinder tube, piston, piston rod, end caps and eye plates, glands, welding, bolting, etc. Additionally, a copy of the designer's calculations (stress and buckling) are also required.

3.6.2 Details of the maximum working, design and test pressures are to be submitted. The maximum push and/or pull forces for which the cylinder has been designed are to be indicated.

3.6.3 A performance curve indicating the maximum permitted push load versus the associated piston rod extension is to be supplied by the manufacturer.

3.6.4 Plans of the hydraulic cylinder are to be submitted together with plans of the lifting or life-saving appliance to which it will be fitted. However, where this is not practicable, details of the appliance to which the hydraulic cylinder will be fitted are to be submitted including the anticipated operating and interface conditions, e.g. dynamic axial forces, type of the appliance, etc.

~~3.6~~ 3.7 Mechanical, electrical and control aspects

~~3.6.1~~ 3.7.1 The following plans are to be submitted for approval, *see also Ch 9 Machinery* and *Ch 10 Electrotechnical Systems*.

(a) Diagrammatic plan of hydraulic or pneumatic systems, where fitted.
(b) Plans of winch gearing, shafts, clutches, brakes, coupling bolts, welded drums, and similar items and their materials and stresses.
(c) Plans of circuit diagram of electrical system, showing load currents and ratings of all electrical equipment, types and sizes of cables, rating type and make of all protecting devices.
(d) Arrangement plan and circuit diagram of switchboard.
(e) General arrangement of control centre.
(f) Schematic diagrams of control panels.
(g) Details of alarms and protection circuits.

~~3.6.2~~ 3.7.2 In addition, the following information is required for reference purposes: calculations of short-circuit currents and main busbars, sub-switchboard busbars and the secondary side of transformers.

Chapter 2
Derrick Systems

■ *Section 2*
Design criteria

2.4 Friction allowance

~~2.4.4~~ 2.4.3 As an alternative, the coefficients of rope tensions may be determined as follows:

$$\cancel{P_{\text{hoisting}}(\mu, ij) = \frac{1}{\sum_{k=0}^{i-1}(1+\mu)^k}(1+\mu)^{i-1}(1+\mu)^{j-1} \quad \text{if } j \neq 0}$$

$$\cancel{= \frac{1}{\sum_{k=0}^{i-1}(1+\mu)^k} \quad \text{otherwise}}$$

$$\cancel{P_{\text{lowering}}(\mu, ij) = \frac{1}{\sum_{k=0}^{i-1}(1+\mu)^k}\,\frac{1}{(1+\mu)^{i-1}}\,\frac{1}{(1+\mu)^{j-1}} \quad \text{if } j \neq 0}$$

$$\cancel{= \frac{1}{\sum_{k=0}^{i-1}(1+\mu)^k} \quad \text{otherwise}}$$

$$P_{\text{hoisting}}(\mu, ij) = \frac{1}{\sum_{k=0}^{i-1}(1+\mu)^k}(1+\mu)^{i-1}(1+\mu)^{j-1} \qquad \text{if } j \neq 0$$

$$= \frac{1}{\sum_{k=0}^{i-1}(1+\mu)^k} \quad \text{otherwise}$$

$$P_{\text{lowering}}(\mu, ij) = \frac{1}{\sum_{k=0}^{i-1}(1+\mu)^k}\,\frac{1}{(1+\mu)^{i-1}}\,\frac{1}{(1+\mu)^{j-1}} \qquad \text{if } j \neq 0$$

$$= \frac{1}{\sum_{k=0}^{i-1}(1+\mu)^k} \quad \text{otherwise}$$

~~2.4.5~~ where

μ = percentage of friction (e.g. 2 per cent for roller bearings)

i = number of parts supporting the load

j = location in drive system (e.g. *see Figure 2.2.1 Coefficients of rope tension* where P_j is defined).

■ *Section 7*
Masts and derrick posts

7.1 General

Table 2.7.1 Minimum thickness of mast plating

Item	Minimum thickness of mast plating, in mm
Curved plates	~~$0{,}32 \frac{d\sqrt{\alpha \sigma_y}}{350 + 2 \text{ x SWL}}$ or $\frac{d\sqrt{\alpha}}{100}$ or $7{,}5$~~ $0{,}32 \frac{d\sqrt{\alpha \sigma_y}}{350 + 2 \text{ x SWL}}$ or $\frac{d\sqrt{\alpha}}{100}$ or 7,5
Flat plates	~~$0{,}32 \frac{b\sqrt{\alpha \sigma_y}}{220 + 2 \text{ x SWL}}$ or $\frac{b\sqrt{\alpha}}{60}$ or $7{,}5$~~ $0{,}32 \frac{b\sqrt{\alpha \sigma_y}}{220 + 2 \text{ x SWL}}$ or $\frac{b\sqrt{\alpha}}{60}$ or 7,5
~~Note~~ **Note** 1. where SWL = the safe working load of the largest derrick operating on the mast, in tonnes *d* = maximum outside diameter of the mast at the position under consideration, in mm. Where the mast is not circular, *d* is to be taken as the maximum diameter of the circle of which the plate forms a part *b* = width of flat plate, in mm, but is to be taken as not less than 60% of the width of the mast at that point measured parallel to the flat plate ~~α = the ratio $\frac{actualtotalstressatthatpoint}{maximumallowablestress}$~~ α = the ratio $\frac{\text{actual total stress at that point}}{\text{maximum allowable stress}}$ ~~Note~~ **Note** 2. Where stiffeners are fitted, *b* may be taken as the mean spacing of stiffeners. The required scantlings of the stiffeners to resist instability under end loading will be considered.	

7.5 Stress calculations – Stayed masts

7.5.5 The bending moment (*BM*) in a single-stayed mast and the tension (*T*) in the stay are to be determined as follows:

BM = (*P* – *T* cosφ) *x*

~~$$= T = \frac{P_{\sec\varphi}}{1 + \left(\frac{3E_p I l_1}{E_s A l^3 cos^2\varphi}\right)}$$~~

$$T = \frac{P_{\sec}\varphi}{1 + \left(\frac{3E_p I l_1}{E_s A l^3 \cos^2 \varphi}\right)}$$

where

φ = the angle of the stay to the horizontal in degrees
A = the cross-sectional area of the stay, in m^2
x = the distance below the mast head about which bending moments are to be calculated, in metres
l = the height of the mast/stay attachment above the deck, in metres
l_1 = the length of the stay, in metres
E_p = the Young's modulus of steel for the mast, in N/m^2
E_s = the Young's modulus of steel for the stay, in N/m^2
I = second moment of area for the mast section, in m^2
P = the component of force acting on the mast head, in Newtons.

Chapter 3
Launch and Recovery Appliances for Survival Craft and Rescue Boats

■ *Section 1*
General

1.6 Safety factors

1.6.2 For steel in which $\sigma_y/\sigma_u \leq 0{,}7$, the allowable compressive stress, σ_a, in primary structural members is given by the following expression:

~~$$\sigma_c = \frac{\sigma_{cr}}{SF \text{ X } \frac{\sigma_y}{\sigma_u}}$$~~

$$\sigma_a = \frac{\sigma_{cr}}{SF \text{ X } \frac{\sigma_y}{\sigma_u}}$$

where

σ_{cr} = critical compressive stress, *see Table 4.2.10 Values of σcr for steel varying σy* in *Chapter 4 Cranes and Submersible Lifting Appliances*

σ_u = minimum ultimate tensile stress, N/mm^2

σ_y = yield stress, N/mm^2

~~≠~~ ~~SF~~ *SF* is given in *Table 3.1.1 Minimum safety factors*.

Chapter 4
Cranes and Submersible Lifting Appliances

■ *Section 2*
Shipboard cranes

2.6 Dynamic forces due to crane movements

2.6.4 In cases where the crane drive control system ensures that motions such as hoisting, travelling or slewing cannot occur simultaneously and the loading caused by one motion is practically zero when the other motion starts, loadings due to hoisting, travelling motions or slewing do not need to be superimposed.

With respect to the load combination formula as given in the Code, this would result in:

Case 1a: $F_h > 1{,}0$ (Hoisting) –> L_{h2}(Travelling or slewing) = 0

Case 1b: ~~$F_h = 1{,}0$ (Live load at rest) –> L_{h2}(Travelling or slewing) > 0~~

$F_h = 1{,}0$ (Live load at rest) –> L_{h2}(Travelling or slewing) > 0

Case 2: Similar.

where

L_{h2} = the next most unfavourable horizontal load (usually due to travelling or slewing acceleration).

2.16 Stability

2.16.1 Travelling cranes, trolleys, grabs, etc. which are capable of travelling whilst loaded are to be examined with regard to stability against overturning for the following conditions:

(a) The worst operating condition as given by load combination Case 2, including forces resulting from an acceleration at deck level of 0,67 m/s^2 or maximum acceleration, if known.

(b) Consideration of sudden release of load in accordance with load combination Case 2, with the hoisting factor, F_h, taken as – ~~0,2~~ –0,2.

2.18 Allowable stress – Compression, torsional and bending members

2.18.2 For members subjected to simple compression, the critical compression stress is given by the Perry-Robertson formulae as follows:

$$\cancel{\sigma_{cr} = \frac{\sigma_y + (1+\eta)\sigma_e}{2} - \sqrt{\left(\frac{\sigma_y + (1+\eta)\sigma_e}{2}\right)2 - (\sigma_y \, \sigma_e)}}$$

$$\sigma_{cr} = \frac{\sigma_y + (1+\eta)\sigma_e}{2} - \sqrt{\left(\frac{\sigma_y + (1+\eta)\sigma_e}{2}\right)^2 - (\sigma_y \, \sigma_e)}$$

where

$$\sigma_e = \cancel{\frac{\pi^2 E}{\left(\frac{KL}{r}\right)^2}} \qquad \sigma_e = \frac{\pi^2 E}{\left(\frac{KL}{r}\right)^2}$$

$$\eta = \cancel{0{,}001a\left(\frac{KL}{r} - 0{,}2\pi\sqrt{\frac{E}{\sigma_y}}\right)} \qquad 0{,}001a\left(\frac{KL}{r} - 0{,}2\pi\sqrt{\frac{E}{\sigma_y}}\right)$$

E = Young's modulus

L = length of member

r = radius of gyration of member

a = Robertson's constant as in *Table 4.2.9 Values of Robertson's constant, a, for various sections*

σ_y = yield stress

K = constant dependent on the end constraint condition of the member and can be obtained from *Table 4.2.8 Value K, for different constraint conditions*.

Values of critical compression stress are given in *Table 4.2.10 Values of σcr for steel for varying σy*. Alternative methods to calculate the simple critical compression stress as per recognised National or International Standards or analysis taking into account second and higher order effects may be considered. In the case where the stability is calculated by means of second or higher order analysis, suitable imperfections are to be taken into account and the loads are to be multiplied by the inverse of the stress factor 1/*F* where the actual stress results are then to be compared with the yield stress of the component.

2.21 Allowable stress – Plate buckling failure

2.21.5 For components subject to combined compression and shear stress, the following allowable stress criteria are to be met:

(a) $\sigma_c \le F\sigma_{cb}$

(b) $\tau \le F\tau_b$

(c) $\cancel{\left(\frac{\sigma_c}{\sigma_{cb}}\right)+\left(\frac{\tau}{\tau_b}\right)^2 \le F}$ $\quad\left(\frac{\sigma_c}{\sigma_{cb}}\right)+\left(\frac{\tau}{\tau_b}\right)^2 \le F$

where

τ = applied shear stress

σ_c = applied compression stress.

2.21.6 For components subject to combined bending and shear stress, the following stress criteria are to be met:

(a) $\sigma_b \le F\sigma_{bb}$

(b) $\tau \le F\tau_b$

(c) $\cancel{\left(\frac{\sigma_b}{\sigma_{bb}}\right)+\left(\frac{\tau}{\tau_b}\right)^2 \le F}$ $\quad\left(\frac{\sigma_b}{\sigma_{bb}}\right)^2+\left(\frac{\tau}{\tau_b}\right)^2 \le F$

where

σ_b = applied bending stress.

2.22 Allowable stress – Buckling failure of thin walled cylinders

2.22.2 For components subject to compression the critical buckling stress is given by:

(a) For $\sigma_{cb}{}^1 < 0,5\sigma_y$

$\sigma_{cb}{}^1 = K^I{}_c E$

(b) For $\sigma_{cb}{}^1 \ge 0,5\sigma_y$

$\cancel{\sigma_{cb}{}^1 = \sigma_y\left(1-\frac{\sigma_y}{4K_c E^I}\right)}$ $\quad\sigma_{cb}{}^1 = \sigma_y\left(1-\frac{\sigma_y}{4K^I{}_c E}\right)$

where

$\sigma_{cb}{}^1$ = critical compressive buckling stress

E = Young’s modulous

r = average radius of tube

t = wall thickness

$K^I{}_c$ = compression buckling constant, *see Figure 4.2.10 compressive buckling constant*.

2.26 Rope safety factors and sheave ratio

2.26.1 The rope safety factor for both running and standing application for cranes with SWL greater than 10 t and less than 160 t is given by:

$$SF = \frac{10^4}{8{,}85\text{SWL} + 1910}$$

where

SF = minimum safety factor required

SWL = safe working load of crane, in tonnes

For cranes with SWL ~~<~~ ≤ 10 t, SF = 5,0
and SWL ~~>~~ ≥ 160 t, SF = 3,0.

This is represented graphically in *Figure 4.2.13 Rope safety factor versus crane SWL*.

The rope safety factor ~~SF~~ *SF* ~~maybe~~ may be reduced for Cases 3 and 4 (*see Ch 4, 2.15 Load combinations 2.15.1* and *Table 4.2.6 Stress factor, F*) by a factor of $\frac{0{,}67}{0{,}85}$.

The safety factor ~~SF~~ *SF* for both the luffing and hoisting system is to be determined using the maximum SWL for each reeving arrangement.

■ *Section 3*
Offshore cranes

3.3 Dynamic forces

3.3.2 The hoisting factor is considered to be dependent on the design operational sea conditions and which is to be defined by the significant wave height and is to be calculated from the following expression:

$$F_{h} = 1 + \frac{v_R}{g}\sqrt{\frac{K}{L_l}}$$

where

K = the crane system stiffness, in N/m
L_l = live load, in kg
~~$v_R = v_{Hs} + \sqrt{v^2{}_D + v^2{}_C}$~~ $v_R = v_{Hs} + \sqrt{v_D{}^2 + v_C{}^2}$ = relative velocity, in m/s
g = 9,91 m/s^2

where

v_D = vertical velocity of load supporting deck (e.g. semisubmersible, vessel, fixed platform, etc.) from which the load is to be lifted
v_C = vertical boom tip velocity at the jib head where the hoisting ropes leave the jib
v_H = minimum hoisting speed (defined in *Ch 4, 3.6 Hoisting speed*)
v_S = hoisting speed
$v_{Hs} = v_H$ in the cases where $0{,}5v_S < v_H$
$v_{Hs} = 0{,}5v_S$ in the cases where $0{,}5v_S \geq v_H$

≠ The load supporting deck velocity v_D and crane jib tip velocity v_C are to be taken from a recognised National or International Standard (e.g. EN 13852) in the absence of project-specific data. The proposed velocities are to be agreed with LR.

The hoisting factor shall not be taken less than that defined in *Ch 4, 2.5 Dynamic forces 2.5.2*, but in no case shall the product of hoisting and duty factor be less than 1,30 for offshore cranes carrying out offboard lifts.

3.6 Hoisting speed

3.6.2 The minimum hoisting speed is to be obtained from the following expression:

~~$v_H = H\sqrt{v^2{}_D + v^2{}_C}$~~ $v_H = H\sqrt{v_D{}^2 + v_C{}^2}$

where

v_H = the minimum hoist speed to avoid re-contact
H = factor to be taken from a recognised National or International Standard (e.g. EN 13852). The proposed factor is to be agreed with LR.

3.7 Slew rings

3.7.5 The ring is also to be considered with respect to fatigue loading based on load combination Case 2 of *Ch 4, 2.15 Load combinations 2.15.1*, multiplied by a load spectrum factor of 0,7 and associated with an allowable stress determined from S-N curves obtained from the type testing of *Ch 4, 3.7 Slew rings 3.7.3* on the basis of 2 x ~~106~~ 10^6 cycles and multiplied by a stress factor = 0,67.

3.9 Rope safety factors

3.9.1 The rope safety factor ~~SF~~ *SF* for offshore cranes is to be determined from the following expression:

$$SF_{swh} = SF\frac{F_{h,swh}}{1{,}6}$$

where

SF_{swh} = safety factor required at significant wave height (swh), but is to be taken as not less than that obtained from *Ch 4, 2.26 Rope safety factors and sheave ratio*
SF = safety factor obtained from *Ch 4, 2.26 Rope safety factors and sheave ratio*
$F_{h,swh}$ = hoisting factor derived in accordance with *Ch 4, 3.3 Dynamic forces*.

≠ The factor $\frac{F_{h,swh}}{1{,}6}$ is not to be taken less than 1,0.

■ *Section 5*
Pedestals and foundation

5.5 Pedestal flange

5.5.1 The bending stress in a pedestal flange (attached to a cylindrical pedestal) may be calculated as follows:

$$\cancel{= \quad \sigma_f = \frac{5\sigma_{total} t_p e}{t^2{}_f} \leq \sigma_a} \qquad \sigma_f = \frac{5\sigma_{total}\, t_p e}{t_f{}^2} \leq \sigma_a$$

where

σ_f = bending stress in flange

σ_{total} = total direct ~~(i.e., bending and tension/compression)~~ stress in pedestal wall below the flange resulting from overturning moment and vertical force

t_f = flange ~~wall~~ thickness ~~(recommendation: $t_f \geq 3t_p$)~~

t_p = pedestal wall thickness below the flange

e = distance between the centre of bolt holes and centre of pedestal wall

The following recommendations shall be taken into consideration:

(a) The thickness of the pedestal wall directly attached to the flange is to be greater than or equal to 1,5 times the minimum theoretical pedestal wall thickness and shall extend over a length defined by 20 per cent of the outside pedestal wall diameter.

(b) The flange thickness is to be greater than or equal to 4,5 times the minimum theoretical pedestal wall thickness.

(c) The distance *e* between the centre of the bolt holes and centre of the pedestal wall is to be less than or equal to 2 times the bolt hole diameter.

(d) The distance between the centre of the bolt holes and the edge of the flange is to be greater than or equal to 1,5 times the bolt hole diameter.

In case the design of the pedestal is not in line with the above recommendations, alternative methods for the assessment of the design of the pedestal may need to be considered.

The allowable stress σ_a is to be taken as defined in *Ch 4, 5.3 Allowable stresses*.

The above formula is intended for initial design calculations of the pedestal flange. In case other effects, e.g. horizontal loading, pre-tension in bolts, size and amount of holes in the flange, reinforcements, plate bending of the pedestal wall ~~tp~~ t_p, etc., are considered to have a significant influence on the pedestal flange design those effects need to be taken into consideration.

Alternative proposals for the calculation of pedestal flanges will be specially considered.

Chapter 5
Shiplift and Transfer Systems

■ *Section 5*
Design loads and combinations

5.7 Allowable stresses

5.7.1 The allowable stress, σ_a, is to be taken as the failure stress of the component concerned, multiplied by a stress factor, *F*, which depends on the load case considered. The allowable stress is given by the general expression:

σ_a = ~~F_σ, or~~ $F\sigma$, or

τ_a = ~~F_τ~~ $F\tau$

where

σ_a = allowable direct stress, in N/mm^2

τ_a = allowable shear stress, in N/mm^2

F = stress factor

σ, τ = failure stress, in N/mm^2.

Chapter 6
Ro-Ro Access Equipment

■ *Section 2*
Loading and design criteria

2.6 Allowable stress – Elastic failure

Table 6.2.1 Stress factor, *F*

Load case	Case 1	Case 2	Case ~~3/4~~ 3
Stress factor, *F*	0,60	0,75	0,85
Note Where an item forms part of the hull structure, the scantlings are to comply with the requirements of the Rules for Ships Case 1: Harbour condition, loading and unloading Case 2: Sea-going condition, loaded in-deck position or stowed unloaded Case 3: Manoeuvring operation or test load			

2.6.8 In case the structural analysis is carried out by means of detailed finite element models, higher allowable stresses can be applied as follows:

(a) $\sigma_{1.FE} \leq 1,1\sigma_a$
(b) $\sigma_{2.FE} \leq 1,1\sigma_a$
(c) $\tau_{o.FE} \leq 1,1\tau_a$
(d) $\sigma_{e.FE} \leq 1,1^2\sigma_a$
~~(e)~~
where
$\boldsymbol{\sigma}_{2.FE}$ = second principal stress
$\tau_{o.FE}$ = shear stress
$\boldsymbol{\sigma}_{e.FE}$ = equivalent stress
$\boldsymbol{\sigma}_{1.FE}$ = first principal stress

Higher allowable stresses, as defined above, may only be applied if the actual stresses are localised. In case the actual stresses can also be calculated by means of analytical methods, the above higher allowable stresses are not applicable and *Ch 6, 2.6 Allowable stress – Elastic failure 2.6.1* are to be applied.

2.12 Hoisting arrangements and items of loose gear

2.12.2 Where wire ropes are used as part of the hoisting arrangement as well as items of loose gear used therein, they are to have a safety factor given by:

$$\cancel{SF = \frac{10^4}{8,85L + 1910}}$$

$$SF = \frac{10^4}{8,8L + 1910}$$

but not less than 4,0 nor greater than 5,0
where
SF = minimum safety factor required
L = safe working load [tonne]

For the calculation of the safety factor, *SF*, only static forces need to be considered, with a friction allowance of 2 per cent for roller bearings.

2.13 Materials

2.13.3 Where the Ro-Ro equipment is subject to certification only, the selected steel grade is to provide adequate assurance against brittle fracture taking into account the material tensile strength and thickness and the environment in which the Ro-Ro equipment is designed to operate and, in general, is to comply with the Charpy test requirements given in *Table 4.2.18 Charpy V-notch impact test temperature requirements for welded primary and secondary steel structure. Excludes stainless steel* to *Table 4.2.20 Charpy V-notch impact test temperature requirements for non-welded components (excluding slew bearings) subject to tensile loading. Excludes stainless steels* in Chapter 4.

Chapter 7
Lifts

■ *Section 2*
Passenger lifts

2.6 Allowable stresses

2.6.1 The allowable stress, σ_a, is to be taken as the failure stress of the component concerned multiplied by a stress factor, *F*, which depends on the load case considered. The allowable stress is given by the general expression:

$\boldsymbol{\sigma}_a$ = ~~$F\sigma$~~ $F\ \sigma$

where

$\boldsymbol{\sigma}_a$ = allowable stress
F = stress factor
$\boldsymbol{\sigma}$ = failure stress.

2.6.8 In the case where the structural analysis is carried out by means of detailed finite element models, higher allowable stresses can be applied as follows:

(a) $\sigma_{xx.FE} \leq 1{,}1\sigma_a$
(b) $\sigma_{yy.FE} \leq 1{,}1\sigma_a$
(c) $\tau_{o.FE} \leq 1{,}1\tau_a$
(d) $\sigma_{e.FE} \leq 1{,}1^2\ \sigma_a$
~~(e)~~ where

$\sigma_{1.FE}$ = first principal stress
$\sigma_{2.FE}$ = second principal stress
$\tau_{o.FE}$ = shear stress
$\sigma_{e.FE}$ = equivalent stress

Higher allowable stresses, as defined above, can only be applied if the actual stresses are localised. In the case where the actual stresses can also be calculated by means of analytical methods, these higher allowable stresses are not applicable and *Ch 7, 2.6 Allowable stresses 2.6.1* are to be applied.

2.16 Emergency means of escape

2.16.3 A trap door in the roof of the lift car with suitable access to it from the inside is to be provided with an opening of at least 0,24 m^2, having a side length not less than 350 mm. Where the lift is solely for passengers, the trap door is to be fitted with a mechanical lock which can only be operated from the outside. Where the lift is solely for crew, the trap door is to be fitted with a mechanical lock which can be operated from inside and outside the car. Alternative emergency evacuation arrangements, procedures or methods instead of a trap door will be specially considered.

Chapter 8
Fittings, Loose Gear and Ropes

■ *Section 2*
Fittings

2.1 Gooseneck and derrick heel assemblies

Table 8.2.4 Stresses in gooseneck and derrick

Item	Boom axial thrust, *T*, in tonnes		
	$T \leq 25$ N/mm^2	~~$25 < T\ 50$~~ $25 < T \leq 50$ N/mm^2	$50 < T$ N/mm^2
Gooseneck pin:			
Bending plus direct stress	90	$40 + 2T$	140
Bearing pressure	$20 + 0{,}5T$	$20 + 0{,}5T$	45
Derrick heel crosspin:			
Shear stress	$25 + 0{,}4T$	35	35
Bending plus shear stress	$90 + T$	$90 + T$	140
Bearing pressure	$20 + 0{,}5T$	$20 + 0{,}5T$	45
Gooseneck pin collar:			
Horizontal bearing pressure	10 N/mm^2		
Minimum diameter	$1{,}15d_1$mm		
Bearing bracket	Total stress in any part is not to exceed $0{,}45\sigma_y$		

2.3 Fixed eyeplates

~~2.3.3~~ 2.3.2 The dimensional details of the fittings may differ at opposite ends depending on the loads to be carried. Where the fitting is made continuous and of the larger thickness required by the Table, care is to be taken to ensure that this thickness is suitable for the proposed shackle or other attachment to the eyeplate.

~~2.3.4~~ 2.3.3 It is highlighted that an increase in the dimension e1 or e2 will result in an increased bending moment on the derrick boom and this may result in increased scantlings.

~~2.3.5~~ 2.3.4 Fixed eyeplates attached to the ship's structure for use with the cargo gear are to have dimensions generally in accordance with *Table 8.2.9 Dimensions of eyeplates at ship's*. Attention is to be given to the stresses which may arise from applied forces not in the plane of the eyeplate. Where the dimensions of the eyeplate differ from the Table values, the safe working load may be taken as:

~~$\mathrm{SWL} = \dfrac{0,04d^2t^2}{4d(b+0,5d)+t(a+0,4d)}$ tonnes~~

$$\mathrm{SWL} = \frac{0{,}04d^2t^2}{4d\left(b+0{,}5d\right)+t\left(a+0{,}4d\right)}\ \text{tonnes}$$

where
dimensions *a*, *b*, *d* and *t* are illustrated in *Figure 8.2.6 Fixed eyeplate at the ship's structure*.

Where the cross-section of the eyeplate varies, the minimum value of (*d* x *t*) is to be used for the calculation.

~~2.3.7~~ 2.3.5 Adequate support is to be provided by the ship structure in way of the eyeplate. Arrangements to give effective spread of the load into the surrounding structure may be required, *see also Ch 2, 8.9 Deck eyeplates*.

■ *Section 3*
Blocks

3.5 Hook blocks

3.5.1 Blocks that are integrated with a hook are known as hook blocks. As an alternative to the allowable stresses given in *Table 8.3.3 Allowable stresses in blocks*, the hook blocks are to comply with all the requirements below:

(a) The hook blocks are to be designed with a safety factor against the ultimate tensile strength as given below:
For hook blocks with SWL ~~<~~ ≤ 25 t, ~~SF~~ *SF* = 5,0
and SWL ~~>~~ ≥ 160 t, ~~SF~~ *SF* = 3,0.
For hook blocks with a SWL between 25 t and 160 t, the safety factor should be based on the equation below:

$$SF = \frac{10^4}{9{,}88\mathrm{SWL} + 1753}$$

where
SF = minimum safety factor required
~~*SWL*~~ SWL= safe working load of hook block, in tonnes.
The minimum safety factor (*SF*) shall be increased by the ratio of F_h/1,6.

(b) The hook block is to be designed by applying the hoist factor and duty factor appropriate for the situation of operation and for the SWL of the hook and using the allowable stress criteria given in *Ch 4, 2.17 Allowable stress – Elastic failure*, for cases 1 and 2.

(c) The hook block is to be designed for the applicable test load for the hook and using the allowable stress criteria given in *Ch 4, 2.17 Allowable stress – Elastic failure*, for case 3.

Large hook blocks, well in excess of 160 t SWL, will be specially considered.

■ *Section 5*
Loose gear

5.1 Shackles

5.1.5 Where the shackle is not manufactured in accordance with a recognised Standard, the safe working load may be taken as the lowest of the values derived from the following formulae:

Side of body:

~~$\mathrm{SWL} = \left(\dfrac{cd^3{}_1}{2r-a+1{,}2d_1}\right)\left(\dfrac{2r}{2r+0{,}5d_1}\right)$ tonnes~~

$$\mathrm{SWL} = \left(\frac{cd_1^{\,3}}{2r-a+1{,}2d_1}\right)\left(\frac{2r}{2r+0{,}5d_1}\right)\ \text{tonnes}$$

Crown of body:

~~$\mathrm{SWL} = \left(\dfrac{cd^3{}_1}{a+d}\right)\left(\dfrac{2r+d_1}{2r+0{,}4d_1}\right)$ tonnes~~

$$\mathrm{SWL} = \left(\frac{cd_1^{\,3}}{a+d}\right)\left(\frac{2r+d_1}{2r+0{,}4d_1}\right)\ \text{tonnes}$$

Shackle pin

~~SWL = $\frac{cd^{3}{}_{2}}{a + d_1}$ tonnes~~ SWL = $\frac{cd_2^{\,3}}{a + d_1}$ tonnes

where all dimensions are in millimetres and are illustrated in *Figure 8.5.1 Shackles*. The value of c is given in *Table 8.5.3 Values of* c *for shackles*.

5.2 Hooks

5.2.3 Hooks are to be forged from killed steel. Forged hooks can be normalised mild steel or suitably heat treated higher strength steel. Cast hooks are not generally permitted, but special consideration ~~will be given to large cast hooks that cannot be manufactured by the forging process.~~ will be given on a case-by-case basis to proposals for cast hooks that are manufactured in accordance with recognised National or International standards. Stress and fatigue calculations supporting the proposal shall be provided.

5.3 Swivels and lifting eyes

Table 8.5.12 Swivels and eyes

Item	Safe working load, in tonnes
Swivel bow piece	$\left(\frac{cd_1^{\,3}}{a + 0{,}4d_1}\right)\left(1{,}75 + \frac{a + d_1}{b + d_1}\right)$ where *b* < 2,55d_1, this value is to be multiplied by 0,22 $\left(2 + \frac{b}{d_1}\right)$
Round eye	$\frac{cd_2^{\,3}}{d_3 + 0{,}4d_2}$ where d_3 < 2,55d_2, this value is to be multiplied by 0,22 $\left(\frac{d_3}{d_2} + 2\right)$
Oval eye	~~$\frac{cd_4^{\,2}}{g + 0{,}4d_4}$~~ $\frac{cd_4^{\,3}}{g + 0{,}4d_4}$ where *b* < 2,55d_4, this value is to be multiplied by 0,22 $\left(\frac{b}{d_4} + 2\right)$
Triangular eye	Top 0,0069*e f* ~~Side $\frac{0{,}0138h\ g2K}{m}$~~ Side $\frac{0{,}0138h\ g^2K}{m}$ ~~Bott om $\frac{0{,}0138k\ j2K}{m}$~~ Bottom $\frac{0{,}0138k\ j^2K}{m}$
Lugs	0,0125*b* ($d_3 - d_2$)
Shank	$c\,d_1^{\,2}$

Note 1. All dimensions are given in millimetres and are illustrated in *Figure 8.5.3 Bow piece for swivel*.
Note 2. Values of *c* and *K* are given in *Table 8.5.13 Values of c for swivel and eyes* and *Table 8.5.14 Form factors, K* respectively.

Chapter 9
Machinery

■ *Section 5*
Hydraulic cylinders

5.1 General

5.1.1 This Section applies to hydraulic cylinders fitted to lifting or life-saving appliances. A typical example is shown in *Figure 9.5.1 Hydraulic cylinder (simplified)*.

5.1.2 For the purposes of certification or classification of lifting appliances, hydraulic cylinders will be considered for structural adequacy including the effects of hydraulic pressure. Although they are not considered to be pressure vessels, they will nevertheless be required to undergo a pressure load test at the manufacturer's works. *See 5.9 Testing* for testing requirements.

5.1.3 Hydraulic cylinders with special features, such as telescopic types (including versions with intermediate supports), self-locking types, with trunnion mounts, with unconventional or no eye plates at one or both ends, etc. shall, as far as possible, be designed in accordance with the requirements outlined in this Section. In such cases, design calculations covering those special features will be required to be submitted.

5.1.4 Where hydraulic cylinders are used in applications covered by specialist Chapters in this Code, requirements given in those Chapters relating to hydraulic cylinders shall also be taken into consideration as applicable.

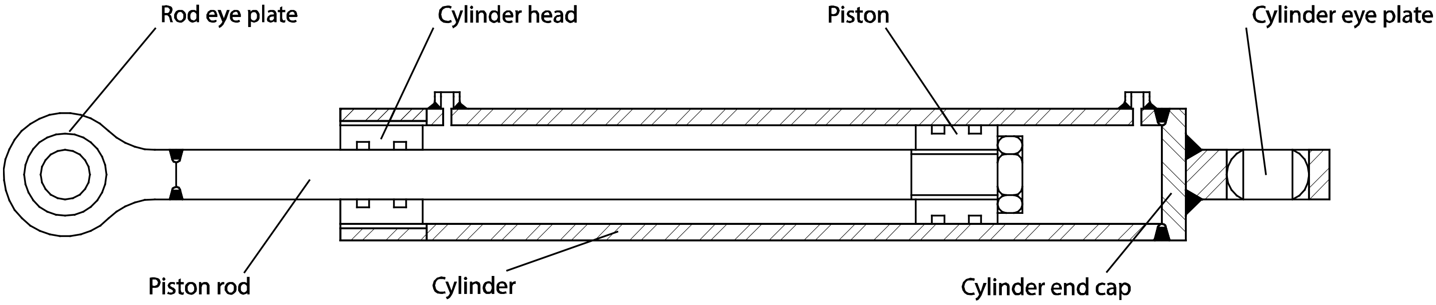

Figure 9.5.1 Hydraulic cylinder (simplified)

5.2 Definitions

5.2.1 Design pressure
The design pressure, P_D is the maximum internal pressure for which the cylinder has been designed. The design pressure is not to be less than the highest set pressure of any relief valves in the hydraulic system and/or relief valves mounted directly on the hydraulic cylinder ports.

5.2.2 Working pressure
The maximum working pressure, P_w, is the maximum dynamic internal pressure that the cylinder may be subject to in operation which includes internal pressures generated by external forces e.g. due to the SWL enhanced by the hoisting factor, accelerations etc., but excludes short duration pressure peaks (*See Ch 9, 5.3 General design requirements 5.3.6*). The maximum working pressure is not to exceed the design pressure, P_D.

5.2.3 Relief valve pressure setting
The relief valve setting is the pressure at which the relief valve is set to activate.

5.3 General design requirements

5.3.1 Welded cylinder pipes should not generally be used for hydraulic cylinder applications. Proposals to use welded cylinder pipes will be specially considered and need to be agreed with LR. The weld procedure including any heat treatment and non-destructive examination shall be submitted to LR for appraisal.

5.3.2 The cylinder head (or gland) shall have adequate length and capability to guide the piston rod under load including the self-weight of the hydraulic cylinder. The seals are to be adequate to prevent hydraulic leakage, and water and dirt entering the cylinder.

5.3.3 The piston rod shall be suitably finished and coated to ensure adequate sealing capability and corrosion protection.

5.3.4 The eye plates shall be equipped, where possible, with spherical bearings to prevent additional end bending moments being introduced. The design shall not allow free movement of the eye plate between the supporting plates in the direction of the pin. Alternative arrangements will be specially considered.

5.3.5 The relief valve pressure setting shall not exceed the design pressure. This setting may be temporarily increased to allow the crane to be proof load tested, but is to be reset for normal operations. *See Ch 9, 5.9 Testing 5.9.5*. To prevent unnecessary activation of the relief valve, it is desirable that there should be a margin between the normal pressure at which the hydraulic cylinder operates and the lowest pressure at which any relief valve is set to lift. The relief valve setting may be up to 1,25 times the maximum working pressure. Alternative proposals will be specially considered.

5.3.6 Short duration pressure peaks which exceed the maximum working pressure, P_w, typical of certain offshore crane applications, need not be considered in determination of the maximum working pressure providing the pressure peaks are less than 0,94 x Hydraulic Test Pressure, P_T (*see Ch 9, 5.9 Testing 5.9.1*) and less than or equal to the Design Pressure, P_D.

5.3.7 The certification requirements for hydraulic cylinders as part of certified lifting and life-saving appliances are detailed in *Table 13.2.1 Minimum requirements for the certification of lifting appliances*. The classification requirements for hydraulic cylinders as part of classed lifting appliances are detailed in *Table 13.3.1 Minimum requirements for the classification of lifting appliances*.

5.4 Design criteria

5.4.1 Hydraulic cylinders fitted to lifting appliances are to be designed to withstand the load combinations as given in the Chapter relevant to the actual application (*see Ch 4, 2.15 Load combinations* and *Ch 4, 3.5 Load combinations* for Shipboard and Offshore cranes, *Ch 7, 2.5 Load combinations* for Lifts). The factored forces (hoisting and duty factor, as applicable), stowage, test load and any exceptional conditions resulting from the actual application shall be considered.

5.4.2 Hydraulic cylinders fitted to life-saving appliances are to be designed to meet the operational and load test conditions required by SOLAS and the IMO LSA Code as applicable.

5.4.3 Depending on the design of the hydraulic cylinder the following calculations and validations are usually required to be carried out as a minimum:

(a) The eye plates shall be designed for the following conditions, *see Ch 9, 5.8 Proof of strength and stability 5.8.4*:
  - local bending, normal and shear-out stress due to pull forces.
  - bearing stress due to pull and push forces.

(b) The following threaded, bolted and/or welded connections (as applicable) are to be calculated as per *Ch 9, 5.8 Proof of strength and stability 5.8.1*, *Ch 4, 2.23 Allowable stress – Joints and connections* (as applicable) and/or *Ch 9, 5.8 Proof of strength and stability 5.8.2* respectively:
  - connection of cylinder or rod eye plates with the cylinder’s end cap or piston rod.
  - connection of piston with piston rod.
  - connection of cylinder head with cylinder tube.
  - connection of cylinder end cap with cylinder tube.

(c) The thickness of the end cap can be calculated as detailed in *Ch 9, 5.8 Proof of strength and stability 5.8.2*.

(d) The hoop and longitudinal stress of the hydraulic cylinder wall shall be calculated as detailed in *Ch 9, 5.8 Proof of strength and stability 5.8.5*.

(e) Diametrical expansion of cylinders with large bores possibly affecting the sealing of the piston is to be considered.

(f) Bolted connections are to be designed in compliance with *Ch 4, 2.23 Allowable stress – Joints and connections*. Tapped holes are to be in compliance with *Ch 9, 5.8 Proof of strength and stability 5.8.1*. The thread-engaging length is to be at least 1,5 times the major bolt diameter.

(g) Axial forces on piston rod.

(h) External pressure on hollow piston rods shall be taken into consideration as applicable.

(i) The pins supporting the eye plates shall be calculated for shear and bending stresses.

(j) Trunnion mounts (if fitted) to the cylinder tube.

(k) Additional calculations for other components of the hydraulic cylinder may be required depending on the actual design.

5.4.4 For the evaluation of the required design, working and relief valves pressures, it is not required to take the duty factor into account. For all other design calculations, the duty factor is required to be taken into consideration.

5.5 Materials

5.5.1 The material requirements (including the Charpy V-notch test requirements) are to be as per the Chapters applicable to the appliance the hydraulic cylinder will be used in. For lifting appliances the materials for hydraulic cylinders are to be selected to meet the requirements detailed in *Ch 11 Materials and Fabrication* and *Ch 4, 2.25 Materials*. For life-saving appliances the materials for hydraulic cylinders are to be selected to meet the requirements of *Ch 11 Materials and Fabrication* and *Ch 3, 1.11 Materials*.

5.5.2 The materials used for hydraulic cylinder applications are to be sufficiently ductile. Materials that have an elongation of less than 14 per cent shall be specially considered by LR.

5.5.3 Pressure retaining parts of carbon-manganese steel grades are to be delivered in the normalised or hot finished condition, whereas alloy grade steels are to be delivered in the quenched and tempered condition.

5.5.4 The use of stainless steel products in hydraulic cylinder applications is required to be specially considered by LR.

5.5.5 In cases where hydraulic cylinders are fitted to life-saving appliances, attention is drawn to materials where the yield strength is very low compared to the ultimate tensile strength. In such cases, it is particularly important to consider the proof load test case as a design case in order to prevent actual stresses being in excess of the yield strength of the material.

5.5.6 For hydraulic cylinder components subjected to tensile loads (e.g. cylinder end caps on pull cylinders) consideration is to be given to the application of Z-grade material. For the evaluation whether Z-grade materials are to be used in the design the methods and criteria of a recognised National or International Standard can be applied.

5.5.7 The required levels of material certification are given in *Table 13.2.1 Minimum requirements for the certification of lifting appliances* and *Table 13.3.1 Minimum requirements for the classification of lifting appliances* for certified and classed appliances respectively.

5.6 Safety features

5.6.1 Hydraulic cylinders are to be equipped with pilot-operated non-return valves at both the inlet and outlet manifolds to ensure that the cylinders remain in position in the event of a hydraulic failure (e.g. hose/tube rupture). The required pilot-operated non-return valves are to be fitted directly to the cylinder ports. The cylinder ports shall either be an integral part of the cylinder wall or shall be connected by means of full penetration welds. Proposals to use valve types other than pilot-operated non-return valves will be specially considered.

5.6.2 The hydraulic system shall be equipped with relief valves to protect the system against overpressure in the hydraulic circuit originating from the lifting appliance itself or the hydraulic system/pump.

5.7 Allowable stresses

5.7.1 The allowable stresses are to be taken from the relevant chapters depending on the nature of the appliance that the hydraulic cylinder will be integral to (*see Ch 3, 1.6 Safety factors* for life-saving appliances, *Ch 4, 2.17 Allowable stress – Elastic failure* for cranes, *Ch 7, 2.6 Allowable stresses* for lifts).

5.7.2 For hydraulic cylinders subject to push loading a second order buckling analysis is to be carried out unless it can be demonstrated that a simplified analysis would give conservative results.

5.7.3 In simplified buckling calculations, the hydraulic cylinder may be idealised using the assumption that the piston rod extends over the complete length of the hydraulic cylinder. The buckling calculations shall then be carried out using the criteria as stated in *Ch 4, 2.18 Allowable stress – Compression, torsional and bending members* for lifting appliances and *Ch 3, 1.6 Safety factors* for life-saving appliances respectively.

5.7.4 In second order buckling calculations, the hydraulic cylinder shall be idealised taking into account the relevant cross-sections (piston rod and cylinder) and shall be calculated by means of the principles of the 2nd order theory. The calculation shall include all relevant, but at least, the following influences:

(a) Dead weight and angle of the hydraulic cylinder;
(b) at least a sinusoidal pre-deformation of L/300 along the various possible hydraulic cylinder lengths (other types and values of pre-deformations maybe required and will be specially considered);
(c) any hoisting factor and material stress factor to be used to enhance the external load used in the calculation;
(d) the bending stresses (due to 2nd order theory) of the cylinder shall be superimposed with the hoop and radial stress due to internal pressure (reference is made to *Ch 9, 5.8 Proof of strength and stability 5.8.5*);
(e) as this is a 2nd order analysis, all safety (stress factor, hoisting factor, etc.) need to be placed on the load side.

5.7.5 Alternative proposals to demonstrate adequate safety against column buckling will be specially considered.

5.8 Proof of strength and stability

5.8.1 Threaded connections (as opposed to bolted connections) shall be calculated as follows:

$$\tau_{Th} = \frac{\frac{2}{\sqrt{3}} F}{\pi \, d_3 \, l} \le \tau_a$$

where

τ_{Th} = shear stress in thread, in MPa
F = axial force applied to the threaded connection, in N
d_3 = root diameter of the threaded connection, in mm
l = length of the threaded connection, in mm
τ_a = allowable shear stress, in MPa

The hydraulic cylinder tube thickness at the threaded section is to be not less than the required thickness for internal pressure measured at the bottom of the internal thread. Alternatively, detailed calculations for the threaded connection as per a recognised National or International Standard (e.g. EN14359) may be performed and submitted for consideration.

5.8.2 Welded and bolted connections shall be designed based on the principles outlined in *Ch 4, 2.23 Allowable stress – Joints and connections*.

5.8.3 The hydraulic cylinder end cap may be calculated using the formula below:

$$\sigma_{ec} = \frac{6 \, p \, \frac{\left(\frac{d_o + d_i}{4}\right)^2}{16} \left(3 + \upsilon\right)}{t^2} \le \sigma_a$$

where

σ_{ec} = bending stress in the end cap due to internal pressure, in MPa
p = internal pressure acting on the end cap, in MPa

d_o = outer diameter of the cylinder, in mm

d_i = inner diameter of the cylinder, in mm

υ = Poisson’s ratio

t = wall thickness of the end cap, in mm

σ_a = allowable bending stress as per *5.7 Allowable Stresses*, in MPa

The above method outlines a simplified approach which does not take into account possible reinforcements of eye plate connected to the end cap. If the end cap is reinforced and the above stress proof fails, alternative means of analysis may be applied.

5.8.4 The eye plates of the hydraulic cylinders may be calculated using the methods of a recognised National or International Standard.

5.8.5 The hoop, longitudinal and radial stresses in the hydraulic cylinder wall can be calculated using the formulae below**:**

(a) Hoop stress, in MPa

$$\sigma_H = p\frac{\left(\frac{d_o}{x}\right)^2 + 1}{\left(\frac{d_o}{d_i}\right)^2 - 1} \le \sigma_a$$

where

σ_H = cylinder hoop stress, in MPa

p = internal pressure, in MPa

d_o = external cylinder diameter, in mm

d_i = internal cylinder diameter, in mm

x = actual diameter of the hydraulic cylinder wall where stresses are required to be evaluated, in mm

σ_a = allowable stress as per *Ch 9, 5.7 Allowable stresses*, in MPa

(b) Longitudinal (axial) stress, in MPa

$$\sigma_L = p\frac{1}{\left(\frac{d_o}{d_i}\right)^2 - 1} \le \sigma_a$$

where

σ_L = cylinder longitudinal (axial) stress, in MPa

(c) Radial stress, in MPa

$$\sigma_R = -p\frac{\left(\frac{d_o}{x}\right)^2 - 1}{\left(\frac{d_o}{d_i}\right)^2 - 1} \le \sigma_a$$

where

σ_R = Radial stress, in MPa

5.8.6 Equivalent stress

The equivalent stress shall be calculated taking into account the following stresses:

(a) Cylinder tube:
- Hoop stress (due to internal pressure).
- Radial stress (due to internal pressure).
- Axial stress (due to external force and/or internal pressure).
- Bending stress (due to 2nd order bending, *see Ch 9, 5.7 Allowable stresses 5.7.4*).

(b) Piston rod:
- Hoop stress (due to external pressure in case of a hollow piston).
- Radial stress (due to external pressure in case of a hollow piston).
- Axial stress (due to external force).
- Bending stress (due to 2nd order bending, *see Ch 9, 5.7 Allowable stresses 5.7.4*).

5.8.7 Where appropriate, fatigue calculations are to be carried out in accordance with a recognised National or International Standard using load cycles and a load spectrum agreed between the manufacturer and the Owner.

5.8.8 The proof of resistance against column buckling is to be carried out as per *Ch 9, 5.7 Allowable stresses 5.7.2*.

5.8.9 Cases where the piston is in contact with the end cap or the cylinder head shall be taken into consideration for the design of the hydraulic cylinder. This is of particular importance for cases where the push or pull forces for the contact situation are above those which can be generated by design internal pressures.

5.9 Testing

5.9.1 The hydraulic cylinder is to be hydraulically tested with the maximum of the following defined test pressures p_T :

(a) $p_T = 1{,}5\ p_W$

(b) $p_T = 1{,}3\ p_D$

where

p_T = hydraulic test pressure, in MPa

p_W = maximum working pressure, in MPa

p_D = maximum design pressure, in MPa

5.9.2 The minimum hydraulic test pressure for life-saving appliances shall be the pressure in the hydraulic cylinder required for the prototype test of 2,2 times the SWL amplified by an additional safety factor of 1,06. Hydraulic test pressures for life-saving appliances used in offshore applications are often dependant on national administrations' requirements and will be specially considered.

5.9.3 The hydraulic cylinders are also to be tested as part of the proof load testing of the lifting or life-saving appliances.

5.9.4 The designer shall ensure that the hydraulic cylinder design is able to withstand the hydraulic test pressure as defined in 5.9.1 and the proof load testing of the lifting or life-saving appliance. The proof loads for lifting appliances are defined in *Ch 12, 1.5 Derricks and derrick cranes*. The proof loads for life-saving appliances are defined in *Ch 3, 1.12 Testing*. The actual stresses due to the applied test pressure and due to proof load testing of the hydraulic cylinders shall not exceed the allowable stresses stated in the relevant chapters applicable to the appliance being considered (*see Ch 4, 2.17 Allowable stress – Elastic failure* for the test loadcase (case 4) for both lifting and life-saving appliances, *Ch 6, 5.2 Load combinations 5.2.4* for Ro-Ro access equipment, etc.).

5.9.5 After proof load testing of the lifting or life-saving appliance, the pressure relief valve is to be reset by the manufacturer to the appropriate operational setting to ensure the appliance cannot be overloaded.

5.10 Handling of personnel

5.10.1 Luffing, folding and telescoping motion shall be realised by two independent hydraulic cylinders where each cylinder is able to hold the SWL for handling of personnel. Alternative arrangements using one hydraulic cylinder will be specially considered.

■ *Section ~~5~~ 6*
Manufacture and testing of machinery

~~5.1~~ 6.1 Materials

~~5.1.1~~ 6.1.1 Materials used in the manufacture of winch machinery are to be in accordance with an internationally recognised Standard. Manufacturer's documentation indicating compliance with a recognised Code or Standard will be accepted.

~~5.1.2~~ 6.1.2 Materials for use in low temperature or hazardous environments are to be in accordance with a National or International Standard acceptable to LR. The Standard used is to be identified in the submitted specification.

~~5.2~~ 6.2 Testing

~~5.2.1~~ 6.2.1 Testing of lifting appliances is to be in accordance with *Ch 12 Testing, Marking and Surveys*.

Chapter 12
Testing, Marking and Surveys

■ *Section 1*
Testing

1.1 General

~~1.1.2 Every item of loose gear is to be proof tested and thoroughly examined before being taken into use for the first time or after any subsequent repair or alteration which may affect the strength of the item.~~

~~1.1.3~~ 1.1.2 Where testing machines (e.g. dynamometers) are used to apply test loads, they are to be of a type approved by Lloyd's Register (LR) as suitable for the intended purpose. The machine is to be calibrated biennially by a recognised authority and the accuracy is to be within ±2 per cent. Testing machines are not to be used for initial testing; and although acceptable to LR for periodical re-testing or following repairs, some National Authorities will not accept this form of testing.

~~1.1.4~~ 1.1.3 Where test weights are used to apply test loads, the weights are to be certified as accurate to within 2 per cent.

~~1.1.5~~ 1.1.4 Water bags may be used instead of certified weights for testing lifting appliances. Where used, these are to be of a type suitable and certified for the purpose. The volume of water is to be measured using a calibrated flow meter which has been certified as accurate to within ±2 per cent. As an alternative, use of calibrated load cells certified as accurate to within ±2 per cent for determining the test loads could also be accepted. The flow meters and load cells are to be re-certified biennially.

~~1.1.6~~ 1.1.5 Suitable precautions are to be taken before commencing the test to ensure the stability of the ship and the adequacy of the supporting structure to bear the test loads.

~~1.1.7~~ 1.1.6 Measures are to be taken to ensure that the appliance can be controlled during the test and to avoid injury or damage which might occur in the event of failure under load.

~~1.1.8~~ 1.1.7 Lifting appliances are generally to hold the test weights for at least 5 minutes, but this may be extended at the discretion of the Surveyor.

1.2 Loose gear

1.2.2 Every item of loose gear is to be proof load tested and thoroughly examined before being taken into use for the first time or after any subsequent repair or alteration which may affect the strength of the item. The proof load applied to each item of loose gear is to be as required by *Table 12.1.1 Proof loads for loose gear* and associated Notes, and illustrated in *Figure 12.1.1 Proof loads for loose gear*.

~~1.2.2~~ 1.2.3 For all blocks (single and multi-sheave), the proof load is to be taken as the resultant load and applied to the head fitting of the block during the test. Where the block is fitted with a becket, the load applied to the becket during the load test of the block will be accepted as the proof test on the becket.

~~1.2.3~~ 1.2.4 Sheave blocks that are permanently attached to, or integral with, the hook are called cargo hook blocks and are to be tested with a proof load for multi-sheave blocks as indicated in *Table 12.1.1 Proof loads for loose gear*. The hooks are to be tested with a proof load for hooks as indicated in *Table 12.1.1 Proof loads for loose gear*.

~~1.2.4~~ 1.2.5 After proof testing, all parts of the blocks are to be thoroughly examined for deformations, cracks, flaws, or other defects and to check that head fittings can swivel and sheaves rotate freely.

~~1.2.5~~ 1.2.6 The proof load may be applied to a Ramshorn hook as indicated in *Figure 12.1.2 Testing of Ramshorn hooks*, but in the latter case an additional load of half the proof load is subsequently to be applied as in *Figure 12.1.2 Testing of Ramshorn hooks*.

~~1.2.6~~ 1.2.7 Short and long link chain is to be subjected to a breaking test in addition to the proof test required by *Table 12.1.1 Proof loads for loose gear*. One sample of length 910 mm is to be taken from each length of chain measuring 185 m or less and is to withstand a breaking load of 4 x SWL for the chain.

~~1.2.7~~ 1.2.8 Where the design of a lifting beam, frame, or similar item is such that the load can be lifted and supported in more than one manner, each arrangement is to be separately tested. Alternative testing proposals outlining how all components will be loaded above their maximum design load (but not above the required test load) will be specially considered subject to the Surveyor's satisfaction. Hooks, shackles and blocks forming part of the lifting beam or frame are to be separately tested in accordance with *Table 12.1.1 Proof loads for loose gear*.

~~1.2.8~~ 1.2.9 Where the loose gear is for use in an offshore, open sea or diving application, the selection of the component of loose gear should take account of the higher proof loads required by Notes 3 and 4 of *Table 12.1.1 Proof loads for loose gear*.

Table 12.1.1 Proof loads for loose gear

Item	Proof load, in tonnes
Single sheave block	4 x SWL
Multi-sheave blocks:	
SWL ≤ 25 t	2 x SWL
25 t < SWL ≤ 160 t	(0,933 x SWL) + 27
160 t < SWL	1,1 xSWL
Hooks, shackles, chains, rings, swivels, etc:	
SWL < 25 t	2 x SWL
25 t < SWL	(1,22 x SWL) + 20
Lifting beams, spreaders, frames, grabs	
SWL ≤ 10 t	2 x SWL
10 t < SWL ≤ 160 t	(1,04 x SWL) + 9,6
160 t < SWL	1,1 x SWL

Note *1*. The safe working load for a single sheave block, including single sheave blocks with beckets, is to be taken as one half of the resultant load on the head fitting.

Note *2*. The safe working load for a multi-sheave block is to be taken as the resultant load on the head fitting.

Note *3*. Where the item is to be used in diving operations, the proof load is to be 1,5 times the proof load value given above for the particular item.

Note *4*. Where the item is to be used for offshore use, the proof loads indicated are to be increased by the ratio $F_h/1,6$ where F_h is derived from *Ch 4, 3.3 Dynamic forces*.

Note *5.* Single sheave blocks that have a resultant load greater than 25 t can have a reduced test load calculated on the basis of a multi-sheave block [(0,933 x RL) + 27 t].

Note *6*. Proof loads are shown graphically in *Figure 12.1.1 Proof loads for loose gear*.

Note *7*.Items not covered above will be specially considered.

■ *Section 3*
Survey requirements

3.2 Initial Survey of new installations

3.2.11 When it is intended to build lifting appliances for certification or classification with LR, constructional plans and all necessary particulars relevant to the lifting appliance as detailed in the Code (*see* Ch 1, 3) are to be submitted for approval before the work is commenced. Any subsequent modifications or additions to the scantlings, arrangements or equipment shown on the approved plans are also to be documented and submitted for approval.

Chapter 13
Documentation

■ *Section 1*
General

1.1 Procedure

1.1.1 The procedure and requirements for the issue of certification by Lloyd’s Register (hereinafter referred to as LR) are specified in ~~*Ch 1, 1.1 Application 1.1.2*~~ *Ch 1, 1.2 Certification*.

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Published by Lloyd’s Register Group Limited
Registered office (Reg. no. 08126909)
71 Fenchurch Street, London, EC3M 4BS
United Kingdom

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